VIBRATION SUPPRESSION APPARATUS AND METHOD FOR HYBRID VEHICLE

BACKGROUND OF THE INVENTION:

5   **Field of the invention**

[0001]   The present invention relates to apparatus and method for suppressing vibrations for a hybrid vehicle having, for example, an engine and two motors as drive sources.

10   **Description of the related art**

[0002]   A Japanese Patent Application First Publication No. 2000-217209 published on August 4, 2000 exemplifies a previously proposed vibration suppression apparatus for a vehicle having a motor as

15 the drive source. In the previously proposed vibration suppression apparatus disclosed in the above-described Japanese Patent Application First Publication, an output error between an actual plant and a plant model is inversely calculated (since

20 torque → rotation angle is double integrals, the inverse calculation is double differentiations) and this, the result of inverse calculation, is conditioned (extraction only from a signal having a predetermined frequency band) due to the output error

25 being caused by only an external disturbance and is additively supplies as a correction torque command. In this construction, each of the actual plant and its model is treated as one-degree-of-freedom motion (second order as a state equation).

30 **SUMMARY OF THE INVENTION:**

[0003]   However, in the previously proposed vibration suppressing apparatus disclosed in the above-described Japanese Patent Application First

Publication, the vibration of the actual plant is suppressed as the one-degree-of-freedom motion. Hence, in a case where the number of drive sources are many and motion degrees of freedom of vibrations of the whole power transmission mechanism in such as the hybrid vehicle is two or more, the vibration suppression is not effective, even if the technique of vibration suppression having one degree of freedom is applied.

[0004]   It is herein noted that the meaning of " not effective " is, for example, in a case where a vibration on an output axle torque is tried to be reduced, a residual rotation vibration within an inside of the planetary gear mechanism is left therein so that wears of elements within the planetary gear mechanism are promoted. In addition, if the vibrations within the planetary gear mechanism are tried to be reduced as low as possible, the residual vibrations on an output axle torque are left thereon so that a vehicular comfortablity is not only lost but also the wear of a power transmission element which is located at the downstream position with respect to a final differential gear is promoted.

[0005]   A problem occurs in a durability of components of the planetary gear mechanism. To prevent the durability problem from occurring, a strength (intensity) of the components thereof cannot help being augmented. Thus, this would result in an expensive manufacturing cost. Furthermore, the drive output torque is vibrated so that an unpleasant feeling is given to a vehicle driver and an

unpleasant noise along with a minute vibration from the components is developed.

[0006] It is, therefore, an object of the present invention to provide vibration suppression apparatus and method for a hybrid vehicle which are capable of effectively suppressing the vibrations of a second-degrees-of-freedom in the planetary gear mechanism, thereby the strength of the planetary gear mechanism being decreased without sacrifice of a durability in components of planetary gear mechanism and the manufacturing cost being reduced, and which are capable of reducing unpleasant vibrations and noises of a drive output torque.

[0007] According to a first aspect of the present invention, there is provided a vibration suppression apparatus for a hybrid vehicle, comprising: a main power source; a plurality of auxiliary power sources; a planetary gear mechanism to modify a gear ratio when an output of the main power source is transmitted to a drive output member; and a vibration suppression control section that selects two power sources whose torque controls are enabled to be performed and superposes a vibration suppression control signal onto each of torque commands supplied to the selected two power sources to suppress two-degrees-of-freedom vibrations of the planetary gear mechanism.

[0008] According to a second aspect of the present invention, there is provided A vibration suppression method for a hybrid vehicle, the hybrid vehicle comprising: a main power source; a plurality of auxiliary power sources; and a planetary gear mechanism to modify a gear ratio when an output of

the main power source is transmitted to a drive output member, the vibration suppression method comprising: selecting two power sources whose torque controls are enabled to be performed; and

5 superposing a vibration suppression control signal onto each of torque commands supplied to the selected two power sources to suppress two-degrees-of-freedom vibrations of the planetary gear mechanism.

[0009] It is noted that the main power source

10 corresponds to, for example, the engine or main motor. It is noted that the plurality of auxiliary power sources correspond to, for example, two or more independent motors, and a single motor having a common stator and two rotors, in an appearance, but

15 functionally having two motor functions. It is noted that the planetary gear mechanism corresponds to Ravigneaux (or spelled Ravigneauwx) (type) compound planetary gear train constituted by a planetary gear train having at least four elements and two degrees

20 of freedom, for example, to couple the four elements of the engine, the first motor, the second motor, and the drive output member. It is also noted that " superposing a vibration suppression control signal (damping purpose correction torque) onto each of

25 torque commands supplied to two power sources correspond to additively supplying the damping purpose (vibration suppression control) signal to each of the torque commands supplied to the two power sources. A steady-state component of each of the

30 torque commands is determined from a torque balance of the planetary gear mechanism. The torque balance of the planetary gear mechanism is determined from the speed of each element of the planetary gear

mechanism and the speed of each element is determined from other constraint conditions such as power performance optimization and fuel consumption optimization.

5 [0010] This summary of the invention does not necessarily describe all necessary features so that the invention may also be a sub-combination of these described features.

BRIEF DESCRIPTION OF THE DRAWINGS:

10 [0011] Fig. 1 is a whole system configuration view of a hybrid drive system and its control system for a hybrid vehicle and its control system for a hybrid vehicle to which a vibration suppression apparatus in a first preferred embodiment is applicable.

15 [0012] Fig. 2 is a block diagram representing a vibration suppression control apparatus in the first embodiment according to the present invention.

[0013] Fig. 3 is a control block diagram representing a vibration suppression controller in

20 the vibration suppression apparatus in the first embodiment shown in Fig. 2.

[0014] Fig.4 is a lever diagram of Ravigneaux type compound planetary gear train used in the vibration suppression apparatus in the first

25 embodiment shown in Fig. 2.

[0015] Fig. 5 is a model view of a translation inertia M and rotation inertia J in a case where a planetary gear mechanism is a four-element, two-degrees-of-freedom planetary gear mechanism

30 (transmission) and elements 1, 2, and 4 are power sources and element 3 is a (drive) output member.

[0016] Figs. 6A and 6B are an integrally flowchart representing a flow of the vibration suppression

control operation executed in the vibration
suppression controller in the first embodiment.

[0017]   Fig. 7 is a control block diagram
representing the vibration suppression controller (or
vibration suppression control section) in a case of
the vibration suppression apparatus in a second
preferred embodiment according to the present
invention.

[0018]   Fig. 8 is a control block diagram
representing the vibration suppression control
apparatus in a third preferred embodiment according
to the present invention.

[0019]   Fig. 9 is a vibration model view of a
four-element, two-degrees-of-freedom planetary gear
mechanism (transmission) in a fourth preferred
embodiment according to the present invention.

[0020]   Fig. 10 is a lever diagram representing the
four-element planetary gear mechanism.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS:

[0021]   Reference will hereinafter be made to the
drawings in order to facilitate a better
understanding of the present invention.

[0022]   (First Embodiment)
Fig. 1 shows a whole system configuration view
representing a hybrid drive system and its control
system of a hybrid vehicle to which a vibration
suppression apparatus according to the present
invention in a first preferred embodiment is
applicable.

[0023]   The hybrid drive system, as shown in Fig. 1,
includes: an engine 1; a coaxial multi-layer motor 2;
a Ravigneaux (type) compound planetary gear train 3;
an output gear 4; a counter gear 5; a drive gear 6; a

differential gear 7; drive axles 8 and 8; a motor and gear casing 9; an engine output axle 10; an output axle 11 of a first motor; an output axle 12 of a second motor (MG2); a motor chamber 13; a gear

5   chamber 14; a drive output axle 15; and a clutch 16.

[0024]   Coaxial multi-layer motor 2 is fixed to motor and gear casing 9 and includes: a stator S as a fixture (stationary) armature on which a coil is wound; an outer rotor OR arranged on an outside of

10 stator S and into which a permanent magnet is buried; and an inner rotor IR arranged on an inside of stator S into which another permanent magnet is buried. These elements are arranged coaxially. Hereinbelow, stator S + Outer rotor OR is called first motor MG1

15 and stator S + inner rotor IR is called second motor MG2.

[0025]   Ravigneaux (type) compound planetary gear train 3 includes four rotary elements having a first pinion P1 meshed with each other; a common carrier C

20 supporting a second pinion P2; and a ring gear R meshed with second pinion P2. It is noted that a double pinion (type) planetary gear is constituted by first sun gear S1, first pinion P1, second pinion P2, and ring gear R and a single pinion (type) planetary

25 gear is constituted by second sun gear S2, second pinion P2, and ring gear R.

[0026]   The hybrid drive system links ring gear R and engine output axle 10 via clutch 16, links first sun gear S1 with first motor output axle 11, links

30 second sun gear S2 with second motor output axle 12, and links common carrier C with output gear 4 (Out) via drive output axle 15.   The output rotation and output torque from output gear 4 passes counter gear

5 → drive gear 6 → differential 7 so as to be transmitted from drive axles 8 and 8 to driven wheels (not shown). The control system of the hybrid vehicle will be described below with reference to Fig. 1. In Fig. 1, the control system of the hybrid vehicle includes: an engine controller 21; a throttle valve actuator 22; a motor controller 23; an inverter 24; a battery 25; a hybrid controller 26; an accelerator opening angle sensor 27; a vehicular velocity sensor 28; a motor temperature sensor 29; an engine speed sensor 30; and bidirectional communications lines 31 and 32. Engine controller 21 outputs a command to command a throttle valve actuator 22 to control the engine torque in accordance with a corresponding command from hybrid controller 26. Motor controller 23 outputs a command to inverter 24 to control rotation speed N1 and torque T1 of first motor MG1 and to control rotation speed N2 and torque T2 of second motor MG2 respectively independently of each other. Inverter 24 is connected to a coil of stator S of coaxial multi-layer motor 3 and generates a compound current which is a composite current of both drive currents to be caused to flow into inner rotor IR and into outer rotor OR. A battery 25 is connected to inverter 24. Hybrid controller 26 inputs sensor signals from an accelerator opening angle sensor 27, a vehicle speed sensor 28, a motor temperature sensor 29, and an engine speed sensor 30. Hybrid controller 26 carries out a predetermined calculation processing. Bi-directional communication line 31 serves to connect between hybrid controller 26 and motor controller 23. Bi-directional communication line 32

serves to connect between hybrid controller 26 and engine controller 21.

[0027] Next, Fig. 2 shows a block diagram representing a vibration suppressive control system of the vibration suppression apparatus according to the present invention.

[0028] In Fig. 2, engine 1 has engine output axle (coupling axle) 10 and an engine purpose speed-and-position detector 16 (a displacement measurement section (means)). First motor MG1 (an auxiliary power source) has a first motor purpose speed-and-position detector 17 (the displacement measurement section (means)) and first motor output axle 11 (coupling axle). Second motor MG2 (an auxiliary power source) has second motor output axle 12 (a coupling axle) and a second motor purpose speed-and-position detector 18 (the displacement measurement section). In Fig. 2, a reference numeral 3 denotes a Ravigneaux (type) compound planetary gear train (planetary gear mechanism) having a ring gear (an element) R, a first sun gear S1 (element), a second sun gear S1 (element), and a common carrier C (element). Common carrier C is linked with drive axle 8 (a drive output member) via drive output axle 15 (coupling axle). In Fig. 2, reference numerals 21 denote engine controller, 23a a first motor controller, 23b a second motor controller, and 23 motor controller, 26 denotes hybrid controller, 26a denotes a vibration suppression controller (vibration suppression control section (means)).

[0029] Motor controller 23 includes first motor controller 23a controlling rotation speed N1 and its torque T1 and second motor controller 23b controlling

rotation speed N2 and its torque T2 of second motor 2.
Each speed-and-position detector 16, 17, and 18
observes a vibration state of each element R, S1, and
S2 in Ravigneaux (type) compound planetary gear train
5   3.   These sensor signals are outputted to engine
controller 21, first motor controller 23a, and second
motor controller 23b.   Hybrid controller 26
determines a target drive torque on the basis of the
accelerator opening angle value (APS), the vehicular
10 velocity detection value (Vsp) and a target drive
torque map, determines a target torque by which
engine 1 is shared and another target torque by which
both of first and second motors MG1 and MG2 are
shared, and outputs a target torque command to engine
15 controller 21.   On the other hand, a steady-state
control for first and second motor torques T1 and T2
and a control (variable speed control) for first and
second motor speeds N1 and N2 are carried out by
first and second motor controllers 23a and 23b.
20 [0030]   In the variable speed control, if engine
speed Ne and gear ratio i (= Ne/No) of Ravigneaux
(type) planetary gear train 3 are already known, in a
lever diagram of Ravigneaux (type) compound planetary
gear train 3 shown in Fig. 4, the following balance
25 equations are established.

$$N1 = Ne + \alpha (Ne - No) \text{ --- } (1)$$

$$N2 = No - \beta (Ne - No) \text{ --- } (2)$$

$$To = T1 + T2 + Te \text{ --- } (3)$$

$$N1 \cdot N1 + N2 \cdot T2 = 0 \text{ --- } (4)$$

30 $\alpha T1 + To = (1 + \beta) T2$ --- (5), wherein N1 and T1 denote
rotation speed and torque of first motor MG1, N2 and T2 denote
rotation speed and torque of second motor MG2, α and β denote
gear tooth ratio of planetary gear train 3, No denotes a drive

output axle rotation speed, To denotes an axial torque of drive output axle 15, and Te denotes an engine output torque.

[0031]   Motor operating points (N1, T1, N2, and T2) are calculated using balance equations of (1) through
5   (5) and a command to obtain the motor operating points (N1, T1, N2, and T2) is outputted by the hybrid controller 26.

[0032]   Vibration suppression controller 26a of hybrid controller 26 selects both of motors MG1 and
10   MG2 as torque controllable two power sources from among the power source and superposes the torque command signal for vibration control purpose and superposes a vibration control purpose torque command signal onto a steady state torque command obtaining
15   motor torques T1 and T2 supplied to both motors MG1 and MG2 so as to suppress a two-degrees-of-freedom vibration of Ravigneaux (type) compound planetary gear train 3, obtaining motor torques T1 and T2 supplied to both motors MG1 and MG2. The reason of
20   selecting both motors MG1 and MG2 from among these power sources by which the torque control is possible from among the power sources is that a torque control response of each of both motors MG1 and MG2 is superior to engine 1. Fig. 3 shows a control block
25   diagram representing vibration suppression controller 26a in the first preferred embodiment. Vibration suppression controller 26a, when Ravigneaux (type) compound planetary gear train 3 is called an actual plant (also called, a real plant) and a
30   vibration dynamic model of Ravigneaux (type) compound planetary gear train 3 is called a plant model, inversely calculates a (an external) disturbance torque using an inverse model of the plant model. A

(damping purpose) correction torque which partially or wholly cancels the inversely calculated disturbance torque is additively supplied to first and second motors MG1 and MG2 from among power

5   sources connected to each element of the actual plant, thus performing a control to suppress two-degrees-of-freedom vibrations of the actual plant. In Fig. 3, reference numerals 261 denote an actual displacement calculating section (separation of translation from

10   rotation), 262 denote a displacement separation section (separation of translation from rotation), 263 denote a model displacement section (plant model), 264 denotes a translation vibration calculation section (vibration displacement calculation section),

15   265 denote a rotation vibration calculation section (vibration displacement calculation section), 266 denote an external disturbance torque calculating section (inverse model of plant model), 267 denote a filter processing section (filtering section) to

20   eliminate noises, 268 denote a correction torque calculation section (synthesis of translation and rotation), 269 denotes a first correction torque adding section, and 270 denote a second correction torque adding section. Actual displacement

25   calculating section 261 serves to calculate an actual rectilinear (or translation) displacement and actual rotation displacement at two selected elements S1 and S2 (refer to Fig. 4) of the actual plant 3 on the basis of the displacement measurement values (x1, x2).

30   Displacement separation section 262 inputs a torque acted upon each of four element R, S1, S2, and C and separates the input respective torques into a translation torque total and a rotation torque total.

[0033]   Translation vibration calculating section
264 calculates a translation error between the
translation model displacement from model
displacement calculating section 263 and the
translation actual displacement from actual
displacement calculating section 261 (translation
vibration displacement).

[0034]   Rotation vibration calculating section 265
calculates a rotation error between the rotation
model displacement from the model displacement
calculating section 263 and the rotation actual
displacement from actual displacement calculating
section 261. Disturbance torque calculating section
266 inversely calculates the external disturbance
torques from the translation error and from the
rotation error using the inverse model of the plant
model of disturbance torque calculating section 266.

[0035]   Filter processing section (filtering
section) 267 carries out a filter processing for the
translation disturbance torque from external
disturbance torque calculating section 266 and that
for the rotation disturbance torque therefrom in
order to eliminate noises included in the signals
indicative of the external disturbance torques.
Correction torque calculating section 268 synthesizes
the translation disturbance torque from filter
processing section 267 with the rotation disturbance
torque filter processed value and calculates a sign
inverted vibration suppression purpose (or damping
purpose) correction torque 1 and a vibration
suppression correction torque 2.

[0036]   First correction torque addition section
269 adds vibration suppression purpose correction

torque 1 calculated by correction torque calculating section 268 to element S1 (first motor MG1). Second correction torque adding section 270 adds the calculated vibration suppression purpose correction

5 torque 2 calculated by correction torque calculating section 268 to element S2 (second motor MG2) (refer to Fig. 3).

[0037] Next, an action of the vibration suppression apparatus in the first embodiment

10 according to the present invention will be described below.

[0038] " Conception of the vibration suppression in a case of the present invention "

In a generally available planetary gear

15 mechanism used in an automatic transmission, a speed constraint between mutual elements of the planetary gear mechanism is present. Hence, degrees of freedom of motion are 2, this controls the revolution speed of the output axle and a gear ratio from the main

20 power source to the output axle. At this time, it is widely known that a relationship in a velocity between each element can be represented by a speed diagram called a lever diagram. Ravigneaux type compound planetary gear train 3 adopted in the first

25 preferred embodiment is an example of the planetary gear mechanism having four-element, two-degrees-of-freedom vibration. Fig. 4 shows the lever diagram of Ravigneaux (type) compound planetary gear train 3. The single pinion type planetary gear can draw a

30 lever diagram such that, in a case where, with a carrier stopped, the sun gear is rotated in a normal direction, ring gear R is reversely rotated. In addition, the double pinion (type) planetary gear can

draw another lever diagram such that, in a case where,
with the carrier stopped and the sun gear rotated in
the normal direction, ring gear R is rotated in the
normal direction at a low revolution. Ravigneaux
5 (type) compound planetary gear train 3 is constituted
by first sun gear S1, first pinion P1, second pinion
P2, and ring gear R. Hence, by combining the lever
diagram of the single pinion (type) planetary gear
with that of the double pinion (type) planetary gear,
10 such a lever diagram aligned in an order from a
leftmost end of Fig. 4 is a first sun gear S1 (first
motor MG1), ring gear R (engine 1), common carrier C
(output gear 4), and a second sun gear S2 (second
motor MG2) can be drawn. From among the rotation
15 elements, if rotation speeds N1 and N2 of first sun
gear S1 and second sun gear S2 are determined, the
speeds of the remaining two ring gear R and common
carrier C are determined.

[0039]   Two degrees of velocities can be expressed
20 by independent two velocities or these arbitrary
linear connections. It is easy to understand for the
two-degrees-of-freedom or velocities to be analyzed
into lever's translation (or rectilinear) mode and
rotation mode without a dynamic (or mechanical)
25 interference. In a case where other vehicles are
selected, terms on the dynamic interference are only
generated. In principle, the same result is obtained.
Within a bracket representing the actual plant in Fig.
3, motions and vibrations of two-degrees-of-freedom
30 power transmission mechanism are shown. In this
bracket, two sections, each section representing an
inertia, viz., $1/Ms^2$ and $1/Js^2$ are shown. These
sections of $1/Ms^2$ and $1/Js^2$ indicate that degrees of

freedom on the motion and vibration are two. Fig. 5
shows models of translation inertia and rotation
inertia in a case of four-element, two-degrees-of-
freedom planetary gear mechanism (transmission) and
5   in a case where elements 1, 2, and 4 denote power
sources and element 3 denote an output member. This
model can be applied to a case where a coupling axle
connecting each element of the planetary gear
mechanism to the inertia of the power source is
10   sufficiently rigid (or stiff or robust) in a
frequency range to be controlled and a torsional
vibration between the inertia of power source and the
corresponding one of the elements of the planetary
gear mechanism is not needed to be taken into
15   consideration. In Fig. 5, an inertia of an element
(for example, rotation inertia of element 4 and so
on) denote a total of the element inertia and
associated power source inertia.

[0040]   It is noted that translation inertia M and
20   rotation inertia J are expressed as: $M = J1 + J2 + J3 + J4$ and $J = J1Acg^2 + J2(Acg - a2)^2 + J3(Acg - a3)^2 + J4(Acg - a4)^2$, wherein $Acg = (a2J2 + a3J3 + a4J4)/M$
and torque arms a2, a3, and a4 denote non-dimensional
(dimensionless) values determined from the gear ratio
25   of the planetary gear mechanism.

[0041]   Although phrase of " superposing a
vibration suppression control signal onto each of
torque commands supplied to two power sources"
described in WHAT IS CLAIMED IS:, steady-state
30   torque commands are determined from a torque balance
of the planetary gear mechanism. The torque balance
of the planetary gear mechanism is determined from
the velocity of each element of the planetary gear

mechanism and the velocity of each element is
determined from other constraint conditions such as
power performance optimization and fuel consumption
optimization (refer to the balance equations of (1)

5   through (5) described above).

[0042]   That is to say, a vibration suppression
effect can be achieved by superposing the vibration
suppression control (damping purpose) signal on the
steady state torque commands without modification of

10 the velocities determined according to each element
determined by each kind of optimization.

[0043]   [Vibration Suppression Control Operation]
Figs. 6A and 6B show integrally an operational
flowchart representing a flow of a vibration

15 suppression control operation executed by vibration
suppression controller 26a in the first preferred
embodiment. At a step S1, actual displacement
calculating section 261 (refer to Fig. 3) calculates
translation displacement and rotation displacement of

20 actual plant 3 from at least two measurement values
by means of speed (velocity) and position detectors
17 and 18 of first motor MG1 and second motor MG2
from among the displacements of the respective
elements. Suppose that the measurement values are $x1$

25 and $x2$ (element 1, element 2) and torque arms from
their respective weight centers to their respective
weight centers are a and b (element 1 to whole weight
center and element 2 to whole weight center and
element 1 to whole weight center). At this time, a

30 calculation equation of each of translation
displacement and rotation displacement is expressed
in a matrix equation described in a bracket of step
S1 in Fig. 6A.

That is to say,

$$\begin{bmatrix} translation & displacement \\ rotation & displacement \end{bmatrix} = \begin{bmatrix} 1 & a \\ 1 & -b \end{bmatrix}^{-1} \begin{bmatrix} x1 \\ x2 \end{bmatrix}.$$

[0044]   At a step S2, translation torque total and
rotation torque total are calculated from a torque
acted upon each element at displacement separation
section (separation of translation from rotation) 262
(refer to Fig. 3) and the routine goes to a step S3.
At step S3, model displacement calculating section
(plant model) 263 calculates the translation
displacement and rotation displacement with
translation torque total and rotation torque total as
inputs of the plant model. Each calculation equation
is expressed as follows:

    Translation displacement = double integrals of
(translation torque total/M) with respect to time.

    Rotation displacement = double integrals of
(rotation torque total/J) with respect to time. It is
noted that translation inertia M and rotation inertia
J are described in the above equations and Figs. 3
and 5.

[0045]   At a step S4, translation vibration
calculating section 264 and rotation vibration
calculating section 265 derives differences (errors
or vibration displacements) between each of the
translation displacements of the plant model and the
actual plant and of the rotation displacements
thereof. At a step S5, (external) disturbance
calculating section 266 (refer to Fig. 3) inversely
calculates the difference in the translation
displacement and the difference in the rotation
displacement to their respectively corresponding
disturbance torques (double differentiation). At a

step S6, filter processing section 267 carries out
the filter processing for the disturbance torque
values, viz., the translation disturbance torque and
the rotation disturbance torque from disturbance

5 torque calculating section 267 in order to eliminate
noises included in the signals. At a step S7,
correction torque calculating section 268 inputs
translation disturbance torque filtered value and
synthesizes them to damping torques 1 and 2 for the

10 selected two elements S1 and S2. A calculation
equation on damping torques 1 and 2 is described in a
bracket of step S7 in Fig. 6B. That is to say,

$$\begin{bmatrix} damping & torque & 1 \\ damping & torque & 2 \end{bmatrix} =$$

$$\begin{bmatrix} 1 & p \\ 1 & -q \end{bmatrix}^{-1} \begin{bmatrix} translation & torque \\ rotation & torque \end{bmatrix}, \quad \text{wherein torque}$$

15 arms between the selected two elements and weight
center are p and q. Then, first correction torque
calculating section 269 and second correction torque
calculating section 270 add the synthesized damping
torque 1 and synthesized damping torque 2 to their

20 corresponding torques T1 and T2 of first motor MG1
and second motor MG2.

[0046]   [Vibration Suppression Action Of Power
Transmission Mechanism By Means Of a Comparison With
a Previously Proposed Vibration Control Method]

25   In a case where the vibration of the power
transmission mechanism occurs or may occur with a
high possibility under a state in which the driving
state and torque distribution are determined, a
method in which a motion state of a vibration

30 occurrence is avoided by shifting the torque
vibration, as described in paragraph No. [0007] in

page 3 of a Japanese Patent Application First
Publication No. 2001-315550 published on November 13,
2001 has been adopted. This unnecessarily deviates
from the actual optimum condition or an increase in

5   numbers of the power sources causes an increase in a
redundancy manner so that a solution balancing
between each kind of optimization and a reduction of
vibrations is derived. At any rate, since the
torque distribution and operating point are deviated

10 from a torque distribution and operating point with,
at an initial stage, only the optimization as a
target, the optimization is not achieved. In
consecutive paragraphs of [0009] and [0010] described
in pages (3) of the above-described Japanese Patent

15 Application First Publication No. 2001-315550, such
an example as described above is described. That is
to say, to suppress the vibration of the vehicle, a
method in which a combination of the engine and a
target power of the motor causes a region in which a

20 lock up is carried out to be determined. As described
above, in a case where one or plural power sources
provide vibration sources under a certain condition
in a previously proposed vehicular vibration control
method, the gear ratio is shifted so as to avoid the

25 certain condition, specifically, a certain
combination between the rotation speed (velocity) of
the above-described power source(s) and torque
thereof. In such a method as described above, in a
case where, as a result of a certain optimization

30 between the power source(s) and the torque thereof is
demanded, it is apparent that the vibration cannot be
suppressed while satisfying this demand. That is to
say, in a state where even under any rotation speed

and torque driving state of every power source, a
part or all of the power sources provide the
vibration sources, it becomes necessary to execute
the driving state while suppressing positively the
5   vibrations.

[0047]   Furthermore, since, in such a multi-element,
multi-degree-of-freedom power transmission mechanism
as described above, backlashes caused by clearances
between the respective gears and elastic materials
10   are intentionally or unavoidably present between each
of the power sources and each of the gears, the
planetary gear itself may be considered to provide
complex and non-linear vibration generation source or
vibration amplifier. In such a case as described
15   above, it is necessary not only to avoid a vibration
generation region but also to avoid a vibration
generation-and-amplification region of the whole
power transmission mechanism. Accordingly, the
degree of freedoms on the driving state is limited
20   and a worsening of a target to be optimized such as
the power performance of the vehicle or fuel
consumption (economy) thereof is resulted. In
addition, since the vibration has a wide range of
conditions and it is not practical to avoid all of
25   vibrations inclusively, a light degree of vibration
is left (neglected). Such a minute vibration as
described above gives an ill effect on a life of each
component of the power transmission mechanism so as
to be avoided. Thus, it is necessary to provide
30   means not only for avoiding the vibration state but
also for positively suppressing the vibration.

[0048]   On the other hand, in the vibration
suppression apparatus in the first embodiment, a

positive vibration suppression method is adopted.
That is to say, in this method, such a problem as
will be described below is carried out. An output
error between an actual plant (or real plant) and the
5   plant model is caused by only the disturbance torque
and is inversely calculated (since torque →
translation displacement and rotation displacement
are carried out under the double integrals, the
inverse calculation is double differentiations) and
10   the inversely calculated displacements are added
(superposed) as the correction torque command to the
steady state torque commands T1 and T21 of both of
first and second motors MG1 and MG2 to each of which
the corresponding one of the selected two power
15   sources is coupled. In other words, such a method
that the vibration disturbance torque which provide a
source of the vibration is cancelled by a torque
compensation. In addition, since, in the vibration
suppression apparatus in the first embodiment, the
20   actual plant and its model (plant model) are treated
as two-degrees-of-freedom motions. In the former
previously proposed vibration suppression apparatus
disclosed in the Japanese Patent Application First
Publication No. 2000-217209, the vibration of the
25   actual plant is suppressed as the one-degree-of-
freedom motion. In the formed previously proposed
vibration suppression apparatus, in a case where the
residual vibration of the output axle torque of the
planetary gear mechanism is present (or left thereon),
30   such a problem that a wear of the elements within the
planetary gear mechanism occurs. However, this
problem can be eliminated in the case of the
vibration suppression apparatus in the first

embodiment. In the former previously proposed vibration suppression apparatus disclosed in the same Japanese Patent Application First Publication, in another case where the vibration within the planetary

5   gear mechanism is reduced as low as possible, the vibration on the output axle torque is left. Such problems that a vehicular comfortability is not only lost but also a wear of a power transmission element which is located at a downstream position with

10   respect to a final speed-reduction (final differential gear) unit occur. However, these problems can be eliminated by the use of the vibration suppression apparatus in the first embodiment. Consequently, the vibrations of the two-

15   degrees-of-freedom developed in Ravigneaux (type) compound planetary gear train 3 can effectively be suppressed.

[0049]   Next, advantages of the vibration suppression apparatus in the first embodiment will be

20   described below.

[0050]   (1) In the hybrid vehicle having the main power source, the plurality of auxiliary power sources, and the plurality of auxiliary power sources, and the planetary gear mechanism to modify the gear

25   ratio when an output of the main power source is transmitted to drive output member, two torque controllable first motor MG1 and second motor MG2 from among the power sources coupled to the planetary gear mechanism 3 are selected, the vibration control

30   (suppression) purpose signals are superposed on torque commands T1 and T2, each of the torque commands being supplied to the corresponding one of first motor and second motor MG1 and MG2. Since the

vibration suppression controller 26a to suppress the
two-degrees-of-freedom vibrations in the planetary
gear mechanism is installed, the two-degrees-of-
freedom vibrations can effectively be suppressed.

5 Consequently, without sacrifice of a durability of
the components (elements) of the planetary gear
mechanism, a strength thereof is lowered and a
manufacturing cost thereof can accordingly be reduced.
When the cost can be reduced, the vibration of the

10 drive output torque and unpleasant noises can be
reduced.

[0051]   (2) When the planetary gear mechanism is
set to be the actual (or real) plant and the
vibration dynamic (mechanical) model of the

15 vibrations of the planetary gear mechanism is set to
be the plant model, vibration suppression controller
26a inversely calculates the disturbance torque using
the inverse model of the plant model and adds the
correction torque which cancels a part or whole

20 disturbance torque into two power sources from among
the power sources coupled to each element of the
actual plant to suppress the two degrees-of-freedom
vibration of the actual plant. Hence, an acting
force by which the vibration is caused to be

25 generated from among the forces acted upon the
planetary gear mechanism can be estimated with a high
accuracy using the plant model and an inverse model
of the plant model. Consequently, two-degrees-of-
freedom vibrations in the planetary gear mechanism

30 can effectively be suppressed.

[0052]   (3) Speed-and-position detectors 16, 17,
and 18 are installed which measure the displacements
of the respective elements developed according to the

torque acted upon each element of the actual plant, vibration suppression controller 26a includes: actual displacement calculating section 261 to calculate the actual displacements using the torques acted upon the

5   respective elements and displacement measurement values; model displacement measurement values; model displacement calculating section 263 to calculate model displacement using the torque acted upon the respective elements and plant model; translation

10   vibration calculating section 264 and rotation vibration calculating section 265 to calculate a vibration displacement which is the error between the actual plant displacement and the plant model displacement; disturbance torque calculating section

15   266 which inversely calculates the disturbance torque using the calculated vibration displacement and the inverse model of the plant model; a correction torque calculating section 268 to calculate the correction torque whose sign of the calculated disturbance

20   torque is inverted; and correction torque addition sections 269 and 270 to add the calculated correction torque to the power sources to which the selected two elements are coupled. Therefore, the vibration suppression controller 26a serves as a control damper

25   generating a damping force for Ravigneaux (type) compound planetary gear train 3 which is a power transmission mechanism. The vibrations of Ravigneaux compound planetary gear train 3 can effectively be damped (attenuated).

30   [0053]   (4) Vibration suppression controller 26a selects first motor MG1 and second motor MG2 each of which is superior in the torque control response from among torque controllable three power sources and

superposes the vibration control (suppression
control) purpose signal on the steady state torque
commands T1 and T2 to be supplied to these two
selected first and second motors MG1 and MG2 so that
5   the two-degrees-of-freedom vibrations of the
planetary gear mechanism are effectively suppressed.
Even if a variation in the torque control response of
each power source as an actuator to suppress the
vibration is present, the two-degree-of-freedom
10   vibrations generated on Ravigneaux (type) compound
planetary gear train 3 of the transmission mechanism
can speedily be suppressed.

[0054]   (5) Since the main power source is engine
1, the plurality of auxiliary power sources are two
15   of first and second motors MG1 and MG2, the
planetary gear mechanism is the four-element, two-
degrees-of-freedom planetary gear mechanism expressed
in the lever diagram in which engine 1 and output
gear 4 are aligned between two motors MG1 and MG2
20   (refer to Fig. 4), and vibration suppression
controller 26a superposes the vibration control
(suppression) signal onto steady state torque
commands T1 and T2 to be supplied to the selected two
motors of first and second motors MG1 and MG2
25   disposed on both ends of the lever diagram,
particularly, the rotation mode vibrations from among
the two-degrees-of-freedom vibrations developed on
Ravigneaux (type) compound planetary gear train 3
which is the power transmission mechanism can
30   effectively be suppressed, and the costs of engine
mount damper and motor mount damper can be reduced.
That is to say, it is usual practice that a damper
constituted by a spring and a mass to reduce ripples

of motor output axles 11 and 12 is installed on
engine output axle 10. The rigidities of motor
output axles 11 and 12 are larger than the rigidity
of engine output axle 10. In addition, if Ravigneaux

5 (type) compound planetary gear train 3 is expressed
in the lever diagram, motor output axles 11 and 12 to
suppress the vibrations are coupled to both ends of
the lever diagram. Hence, the vibrations in the
rotation mode of Ravigneaux (type) compound planetary

10 gear train 3 can effectively be suppressed. As a
result of this, without sacrifice of the durability
of the compounds of Ravigneaux (type) compound
planetary gear train 3, the strength thereof can be
decreased and the manufacturing cost can be reduced.

15 In details, since the size and weight of spring and
mass of the damper can be reduced, the cost of the
engine damper can be reduced. Furthermore, since the
variations in the output torque can effectively be
reduced, a necessity of reducing the torque ripples

20 by means of the damper of engine output axle 10 can
be reduced within a range of an impediment of a
smooth rotation of engine 1. In other words, since
the size and weight of spring and/or mass of the
damper can be reduced, the cost of the engine damper

25 (engine mount) can be reduced. The, due to the
minute velocity (speed) vibration and the variation
of the generation torque to correct the minute
velocity (speed) vibration, a magnetic flux within a
motor iron core is resulted in having the ripple

30 thereof. This causes an iron loss within the iron
core to be increased. The vibration suppression
control causes the minute speed vibration to be
reduced. The iron loss can be reduced from the

reduction in vibration of the magnetic flux in the iron core. A thermal energy of the motor is equivalently increased so that a capacity of the motor can nearly fully be used. The increase in the thermal capacity of the motor can be utilized in the decrease in the manufacturing cost.

[0055]   (6) Since the main power source is engine 1 and the plurality of auxiliary power sources are coaxial multi-layer motor 2 having a single stator S and two rotors IR and OR and Ravigneaux (type) compound planetary gear train 3 expressed in the lever diagram in which engine 1 and output gear 4 are aligned in the lever diagram between two motors MG1 and MG2, there are greater advantages in terms of the cost, the size, and the efficiency as compared with a case where two independent motors are adopted and the planetary gear can be compacted in its axial direction as compared with the case where the two independent motors are adopted. Furthermore, a compatibility of a combination of co-axial multi-layer motor 2 with Ravigneaux (type) compound planetary gear train 3 is favorable and can be constituted by a preferable hybrid drive system. That is to say, since two-rotor, one-stator coaxial multi-layer motor 2 is adopted, a current for inner motor IR and a current for outer motor OR are superposed to form a compound current and the compound current is caused to flow through the single stator coil so that two rotors IR and OR can respectively be controlled independently of each other. In details, although, in terms of an outer appearance, this is a single coaxial multi-layer motor 2 and this combination can be used as different or same kind of functions of

motor function and generator function. Hence, as
compared with a case where two independent motors
having the rotors and stators, respectively, are
installed, greater advantages can be obtained in
5   terms of the cost (reduction in number of parts,
reduction in an inverter current rating, and
reduction in magnet number), the size
(miniaturization in terms of coaxial structure, and
reduction in inverter size), and the efficiency
10   (reduction in iron loss and reduction in inverter
loss). In addition, only a control over the compound
current can achieve a usage of not only the motor and
the generator but also the generator and the
generator. In the way described above, a high degree
15   of selection of freedom can be provided. For example,
as described in the first embodiment, in the case
where the coaxial multi-layer 2 is adopted in drive
sources of the hybrid vehicle, a most effective or
most efficient combination can be selected in
20   accordance with the driving condition from among a
multiple number of selections.
[0056]   Ravingneaux (type) compound planetary gear
train 3 achieves the combination of four planetary
gears (two parallel longitudinal planetary gears and
25   two crossing forward-rearward direction planetary
gears) although a width size is two-train planetary
gears. Hence, for example, as compared with an axial
alignment of four planetary gears, an axial
directional size can be shortened. In a case where
30   coaxial multi-layer motor 2 and Ravigneaux (type)
compound planetary gear train 3 are applied to the
hybrid vehicle drive system, since they are mutually
of the coaxial structure, the output axles 11 and 12

of the coaxial multi-layer motor 2 and sun gears S1
and S2 of the co-axial multi-layer motor 2 and sun
gears S1 and S2 of Ravigneaux (type) compound
planetary gear train 3 can simply be linked together

5   by means of, for example, a spline coupling. The
compatibility of the combination is very favorable
(good) and this combination is extremely advantageous
from the standpoints of space, cost, and weight. In
a case where one of coaxial multi-layer motor 2 is

10   used as a discharger (motor) and the other thereof is
used as a generation (generator), it is possible to
control the motor current via single inverter 24. A
discharge from battery 25 can be reduced. For
example, in a case of a direct power distribution

15   control mode in which the balance equations (1)
through (5) described above are established,
theoretically, the discharge from battery 25 can be
zeroed. In a case where both rotors IR and OR of
coaxial multi-layer motor 2 are used as motors

20   together with the single stator S, a range of the
drive of the hybrid vehicle can be widened.
[0057]   (Second Embodiment)
In the first embodiment, the vibration disturbance
torques causing the vibrations are directly cancelled

25   by means of a, so-called, torque compensation method.
However, in the second embodiment, the vibrations
caused by the vibration disturbance torques are
speedily damped (attenuated) by means of a
controllable damping torque method. That is to say,

30   vibration suppression controller 26a, as shown in Fig.
7, includes: actual displacement calculating section
261 that calculates the translation displacement and
the rotation displacement of the actual plant using

the torque acted upon each element in the actual
plant and the displacement measurement values; a
damping torque calculating section 271 that
calculates a translation damping torque and a
5   rotation torque damping torque using the actual
translation and rotation displacements and an
electrical damper (also called an electric damper or
called an attenuator); filter processing (filtering)
section 267 that eliminates noises from the
10   translation damping torque and from the rotation
damping torque; a correction torque calculating
section 268 that synthesizes a filtered value of the
translation damping torque and the filtered value of
the rotation damping torque and calculates a
15   vibration suppression (or damping purpose) correction
torque 1 and a vibration suppression (damping
purpose) correction torque 2, each sign of correction
torques 1 and 2 being inverted with respect to the
translation and rotation displacements; a first
20   correction torque adding section 269 that additively
supplies the vibration suppression (damping purpose)
correction torque 1 to element S1 (first motor MG1);
and a second correction torque adding section 270
that additively supplies the vibration suppression
25   (damping purpose) correction torque 2 to element S2
(second motor MG2). It is noted that since other
structures as described in the first embodiment are
generally the same as those of the second embodiment,
the detailed description thereof will be omitted
30   herein.

[0058]   The action of the second embodiment will be
described below. Actual displacement calculating
section 261 calculates the translation displacement

and the rotation displacement using the torque acted
upon each element of the actual plant (Ravigneaux
type compound planetary gear train 3) and
displacement measurement values. At damping torque
5   calculating section 271, the translation and rotation
damping torques are calculated using the translation
displacement, rotation displacement, and the
electrical (electric) damper. At filtering
(processing) section 267, the noises are eliminated
10 from the translation damping torque and from the
rotation damping torque. At correction torque
calculating section 268, the translation damping
torque filtered value is synthesized to the rotation
damping torque to calculate sign inverted vibration
15 suppression (damping purpose) correction torques 1
and 2. Vibration suppressing correction torque 1 is
additively supplied to element S1 (first motor MG1)
at first correction torque adding section 269 and
vibration suppressing correction torque 2 is
20 additively supplied to element S2 (second motor MG2)
at second correction torque adding section 270.
    [0059]   Next, the advantage that the vibration
suppression apparatus in the second embodiment will
be described below. Since the vibration suppression
25 apparatus for the hybrid vehicle in the second
embodiment, damping torque calculating section 271 is
disposed to calculate the translation damping torque
and the rotation damping torque using the translation
displacement, the rotation displacement, and the
30 electrical (electric) damper, a simple vibration
suppression controller 26a without use of the plant
model can be achieved. In addition, the vibrations
caused by the vibration disturbance torques can

speedily be damped by means of the controllable damping torque (damping purpose correction torques). It is noted that C_Ms and C_Js described in block 271 denote transfer functions of the electrical

5   (electric) damper.

[0060]   (Third Embodiment)

In the second embodiment, the translation and rotation damping torques are determined using the translation displacement, the rotation displacement,

10   and the electrical (electric) damper. However, in a third embodiment of the vibration suppression apparatus according to the present invention, using the errors in the translation and rotation displacements between the actual plant and plant

15   model and the electrical (electric) damper (or attenuator described above), the translation and the rotation damping torques are determined.

[0061]   That is to say, as shown in Fig. 8, the vibration suppression controller 26a in the third

20   embodiment includes: displacement separating section 262 that inputs (receives) torques acted upon respective elements R, S1, S2, and C of Ravigneaux (type) compound planetary gear train 3 and that separates the torques into translation torque total

25   and the rotation torque total; a model displacement calculating section 263 that calculates a translation model displacement and a rotation model displacement at selected two elements S1 and S2 using the translation torque total and the rotation torque

30   total both from displacement separation section 262 and the plant model; a translation vibration calculating section 264 that calculates the error in the translation (a translation vibration causing the

displacement) which is an error between the
translation model displacement from model
displacement calculating section 263 and the actual
translation displacement from actual displacement

5   calculating section 261; and a rotation vibration
calculating section 265 that calculates the error of
the rotation (rotation vibration displacement)
between the rotation model displacement from model
displacement calculating section 263 and the rotation

10 actual displacement from actual displacement
calculating section 261, in addition to the structure
in the case of the second embodiment.

[0062]   The action of the third embodiment will be
described below. A damping torque calculating

15 section 271' calculates the translation damping
torque and the rotation damping torque using the
error in the translation displacement, the error in
the rotation displacement, and the electrical
(electric) damper. Filtering section 267 eliminates

20 noises from the translation damping torque and the
rotation damping torque. Correction torque
calculating section 268 synthesizes the translation
damping torque filtered value and rotation damping
torque filtered value to calculate the sign inverted

25 vibration suppression correction torque 1 and the
sign inverted vibration suppression correction torque
2. First correction torque calculating section 269
additively supplies the vibration suppression
correction torque 1 to element S1 (first motor MG1)

30 and second correction torque calculating section 270
additively supplies the vibration suppression
correction torque 2 to element S2 (second motor MG2).

[0063]   Next, advantage of the third embodiment
will be described below. In the vibration suppression
apparatus for the hybrid vehicle in the third
embodiment, damping torque calculating section 271'
5   to calculate the translation damping torque and the
rotation damping torque using the translation
displacement error described above, the rotation
error described above, and the electrical (electric)
damper is provided. The impediment of the velocity
10   control of the planetary gear mechanism becomes a few
as compared with the vibration suppression control
method executed in the second embodiment.
[0064]   (Fourth Embodiment)
The vibration suppression apparatus in a fourth
15   preferred embodiment according to the present
invention will be described below. In the first,
second, and third embodiments, an example having such
a high rigidity that torsional vibrations between
each power source and its coupling axle and between
20   each element of the planetary gear mechanism are
negligible is supposed. On the other hand, in the
fourth embodiment, such an example that elastic
vibration torsional vibrations between each power
source and its coupling axle and between each element
25   of the planetary gear mechanism are not negligible is
suppressed. Fig. 9 shows a vibration model of a
four-element, two-degrees-of-freedom planetary gear
mechanism (transmission). In the fourth embodiment,
a control torque of one of the power sources
30   connected to the selected two axles having high
rigidities is used to suppress the vibrations. It is
noted that, if translation inertia is denoted by M
and rotation inertia is denoted by J, inertias M and

J can be expressed as follows: $M = J1 + J2 + J3 + J4$, $J = J1Acg^2 + J2(Acg - a2)^2 + J3(Acg - a3)^2 + J4(Acg - a4)^2$, wherein $Acg = (a2J2 + a3J3 + a4J4)/M$ and torque arms a2, a3, and a4 are dimensionless values

5 determined from gear ratio of the planetary gear mechanism. Suppose that power sources of 1 and 4 are selected in the way as described above and with these elastic coupling axles directly coupled, the vibration suppression apparatus can be executed using

10 the control section of the vibration suppression controller 26a shown in each of Figs. 3, 7, and 8. In this case, J1 and J4 shown in Fig. 9 are replaced with J1 + Jm1 and with J4 + Jm4, respectively. That is to say, $M = J1 + Jm1 + J2 + J3 + J4 + Jm4$, $J = (J1 +$

15 $Jm1)Acg^2 + J2(Acg - a2)^2 + J3(Acg - a3)^2 + (J4 + Jm4)(Acg - a4)^2$, wherein $Acg = (a2J2 + a3J3 + a4(J4 + Jm4))/M$. Furthermore, it is necessary to insert such a low pass filter (LPF 261A in each of Figs. 3, 7, and 8) that does not pass the vibration components

20 whose frequencies are equal to or higher than one of the frequencies of resonance frequencies of coupling axles coupling the selected power sources 1 and 4 to the corresponding elements (first element 1 and fourth element 4, refer to Fig. 9) detecting section

25 of the calculated translation inertia and the rotation inertia in each in Figs. 3, 7, and 8. A position of the above-described low pass filter may be placed in front of or behind of the section (for example, 261) in each of Figs. 3, 7, and 8 in which

30 separation of translation from rotation is described. [0065]   Next, the advantages of the vibration suppression apparatus for the hybrid vehicle in the fourth embodiment will be described below. That is

to say, in the vibration suppression apparatus for the hybrid vehicle in the fourth embodiment, vibration suppression controller 26a selects the power sources to which the coupling axles having two

5 high resonance frequencies from among the resonance frequencies on a torsional vibration system coupling between the respective elements of the planetary gear mechanism and each power source, superposes the vibration control purpose signals (damping purpose

10 correction torques 1 and 2) on the torque commands given to the two elements to each of which the corresponding one of the two selected power sources is coupled via the corresponding one of the coupling axles to suppress the two-degrees-of-freedom

15 vibrations of the planetary gear mechanism. Hence, even if the torsional vibrations occur on the coupling axles coupling the power sources as the actuator to suppress the vibration to the respective elements of the planetary gear mechanism, the two-

20 degrees-of-freedom vibrations developed on the planetary gear mechanism can speedily be suppressed up to a highest frequency within a range in which no excitation for the torsional vibration occurs. Consequently, without sacrifice of the durability of

25 the elements of the planetary gear mechanism, the strength (intensity) can be lowered and the manufacturing cost thereof can accordingly be reduced. In addition, the vibrations of the drive output torque and unpleasant noises can be reduced.

30 [0066]   The vibration suppression apparatus for the hybrid vehicle according to the present invention has been described with reference to the first, second, third, and fourth preferred embodiments. A specific

structure is not only limited to the first, second, third, and fourth embodiments. Various changes and modifications may be made without departing from the sprit and the scope of the present invention. In

5 each of the first, second, third, and fourth embodiments, the vibration suppression section is separated into the translation mode and rotation mode. The vibration suppression section may be separated into the displacement of element 1 and that of

10 element 2. In each of the first, the second, the third, and the fourth embodiments of the vibration suppression apparatus according to the present invention, first motor MG1 and second motor MG2 are constituted by coaxial multi-layer motor 2 having

15 common stator S and two rotors IR and OR and functionally achieving two motors although, in appearance, coaxial multi-layer motor 2 is a single motor.

[0067] In each of the first, the second, the third,

20 and the fourth embodiments, the planetary gear mechanism is constituted by, as an application example, Ravigneaux (type) compound planetary gear train 3. However, the planetary gear mechanism is not limited to Ravigneaux (type) planetary gear train

25 3 but may be constituted by the planetary gear having at least four elements and two degrees of freedom to couple the four elements of the engine, the first motor, the second motor, and the output member. That is to say, as shown by a lever diagram of Fig. 10, if

30 any arbitrary two elements' velocities (revolutions per unit time) are determined, the remaining two element's velocities are determined. Or alternatively, if an arbitrary one element's velocity (speed) and a

speed ratio between the arbitrary two elements (for example, if the engine output axle and the transmission output axle are selected, this indicates the gear ratio) are determined, the velocities

5 (speeds) of all elements are determined. This is represented as the four-elements, two-degrees-of-freedom planetary gear mechanism.

[0068]   The entire contents of a Japanese Patent Application No. 2002-245722 (filed in Japan on August

10 26, 2002) are herein incorporated by reference. The scope of the invention is defined with reference to the following claims.

15

20

25

30